9 Cogeneration Plants

9.1 Introduction

In conventional utility power plants, electric energy is generally produced with an overall efficiency, that is, the ratio between useful output and input power as fuel, in the range 35–60 %, because of the large quantity of heat discharged into the atmosphere without recovery through cooling towers, lakes, or rivers.

Differently, in cogeneration plants (heat and power plants), either useful heat and mechanical or electric power are generated from fuel, or power is produced by recovering heat from processes. An overall efficiency ranging from 60 to 85 % can be achieved, depending on the type of cogeneration plant. If useful heat is transformed into cooling media by an absorption system, the plant is called trigeneration plant.

Cogeneration is an effective method of primary energy conservation; from this point of view, cogeneration can be applied whenever it is economically justified. The correct use of this system implies a balance between electric-power requirements and process-heat requirements in quantity and in quality. If this balance does not exist, electric power must be exchanged with utility and boiler plants must produce additional heat. The discharge of unnecessary heat should be avoided; otherwise, the system works with lower efficiency like a conventional utility plant.

Before the widespread diffusion of utilities producing and distributing electric energy to end users, cogeneration plants were very common in industry.

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Afterwards, because of local regulations and tariffications, these plants have been unevenly located and used, the choice depending on the cost of fuel and electric energy tariffs.

9.2 Forms of Cogeneration and Trigeneration

Cogeneration plants can be grouped basically into two types referred to as topping cycles and bottoming cycles.

The topping cycles produce power, mechanical or electric, before delivering thermal energy to the end users. Typical examples are the backpressure or non-condensing steam turbine, the gas turbine and combined cycles, and the reciprocating engine where exhausts are utilized as heat for end-user needs.

The heat can also be used as input into an absorption refrigerating system for cooling.

The bottoming cycle recovers thermal energy, which would normally be discarded, to produce process steam and electricity. In this cycle, first thermal energy is used for the process, and then the exhaust energy is used to produce mechanical or electric power at the bottom of the cycle. This cycle is most attractive where there is a large quantity of thermal energy at a temperature of 623 K (350 °C; 662°F) or greater associated with exothermic reactions, as in many chemical processes and in rotary kilns and furnaces. Recovery of the steam to be used in a steam turbine is the commonest bottoming cycle; for lower temperatures other cycles, like the Rankine cycle with organic fluids (ORC—Organic Rankine Cycle), are sometimes used. In this case, the electric efficiency is around 20 %, and the electric output power ranges from 200 kW to 20 MW and more.

ORC plants are also used as topping cycles, usually fed by solid biomass.

The electric generator may be synchronous or asynchronous; the choice between them depends on the working mode: if the system is independent of the utility grid, the synchronous generator must be used; if the system is interconnected with the utility grid, either type can conveniently be used. The generator voltage can be at low or medium level depending on the size and on the layout of the internal distribution network.

Table 9.1 shows the main typical parameters of cogeneration technologies based on both fossil fuel and biomass.

Figures 9.1 and 9.2 show simplified models of cogeneration and trigeneration plants fed by natural gas, which can also be used in the case of other fuels or biomasses. Electrical and heating efficiency parameters from Table 9.1 are the input together with the rated output electrical power; in trigeneration plants conversion from heating efficiency to cooling one is made by using the absorption system COP (practically less than 0.75).

Table 9.1 Technical parameters of generation power plants fed by natural gas, solid biomass, liquid biomass, and standard coal

Technology	*Electrical efficiency (%)*	*Heat recovery efficiency (%)*	*Fuel*	*DG = distributed generation; UP = utility plant*
Reciprocating engine	35–45 %	45–35 %	Natural gas	DG
Gas turbine	25–35 %	55–45 %	Natural gas	DG
Micro gas turbine	18–20 %	62–60 %	Natural gas	DG
Boiler and backpressure steam turbine	12–20 %	68–60 %	Natural gas	DG
Boiler and backpressure steam turbine	15 %	65 %	Solid Biomass	DG
Reciprocating engine	35–40 %	45–40 %	Liquid Biomass	DG
Utility plant combined cycle 800–1,000 MW	60 %	0	Natural gas	UP
Utility plant, steam condensing turbine	26 %	0	Coal	UP

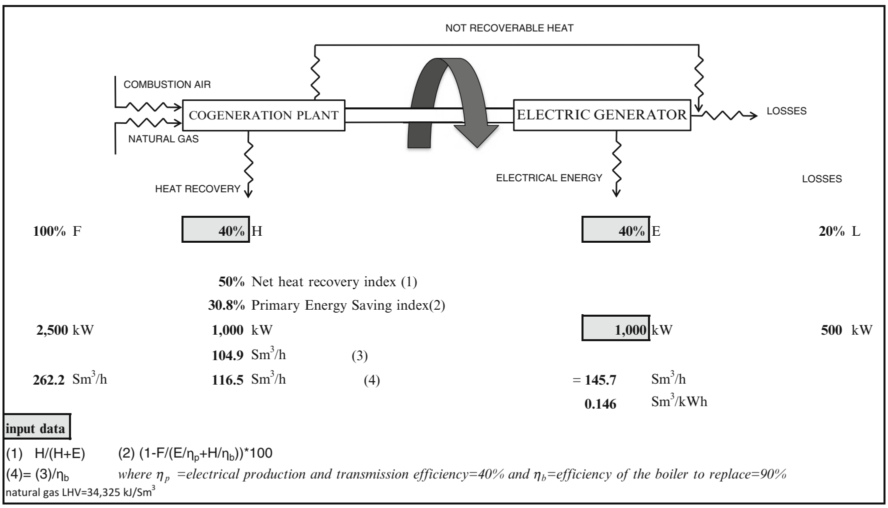


Fig. 9.1 Basic scheme of a cogeneration plant

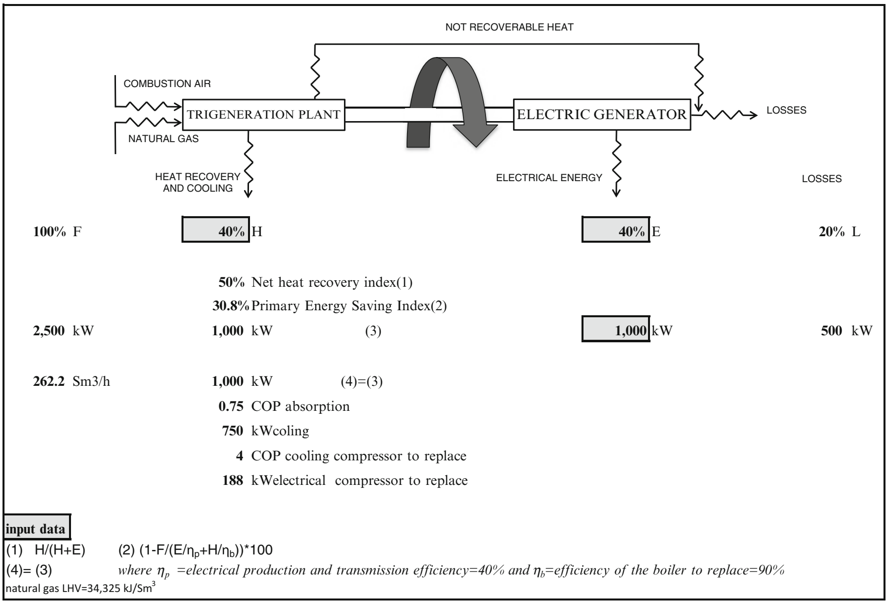


Fig. 9.2 Basic scheme of a trigeneration plant

9.3 The Backpressure or Non-condensing Steam Turbine

Figure 9.1 represents a cogeneration plant with backpressure steam turbine exhausting steam headers to the plant process. The efficiency coefficients are those from Table 9.1. Figure 9.1 may also represent a utility plant designed to generate electric power if appropriate coefficients are used.

The amount of power that can be produced by expanding steam in a prime mover is limited by the Available Energy (AE) between the inlet and outlet of the steam turbine. This energy is the enthalpy difference between the inlet superheated steam, at high pressure and temperature, and the outlet steam at lower pressure along an ideal isentropic expansion. The Mollier diagram or equivalent steam tables (see Sect. 6.4) can conveniently be used for this purpose (see Fig. 9.3). Alternatively, Theoretical Steam Rate tables such as those published by ASME can be used; these report the Theoretical Steam flow Rate (TSR) required to generate 1 kWh in a 100 % efficiency expansion process (see Table 9.2). TSR is the ratio between the energy content of 1 kWh (3,600 kJ/kWh) and the Available Energy AE (kJ/kg); it represents the amount of steam theoretically needed to produce 1 kWh:

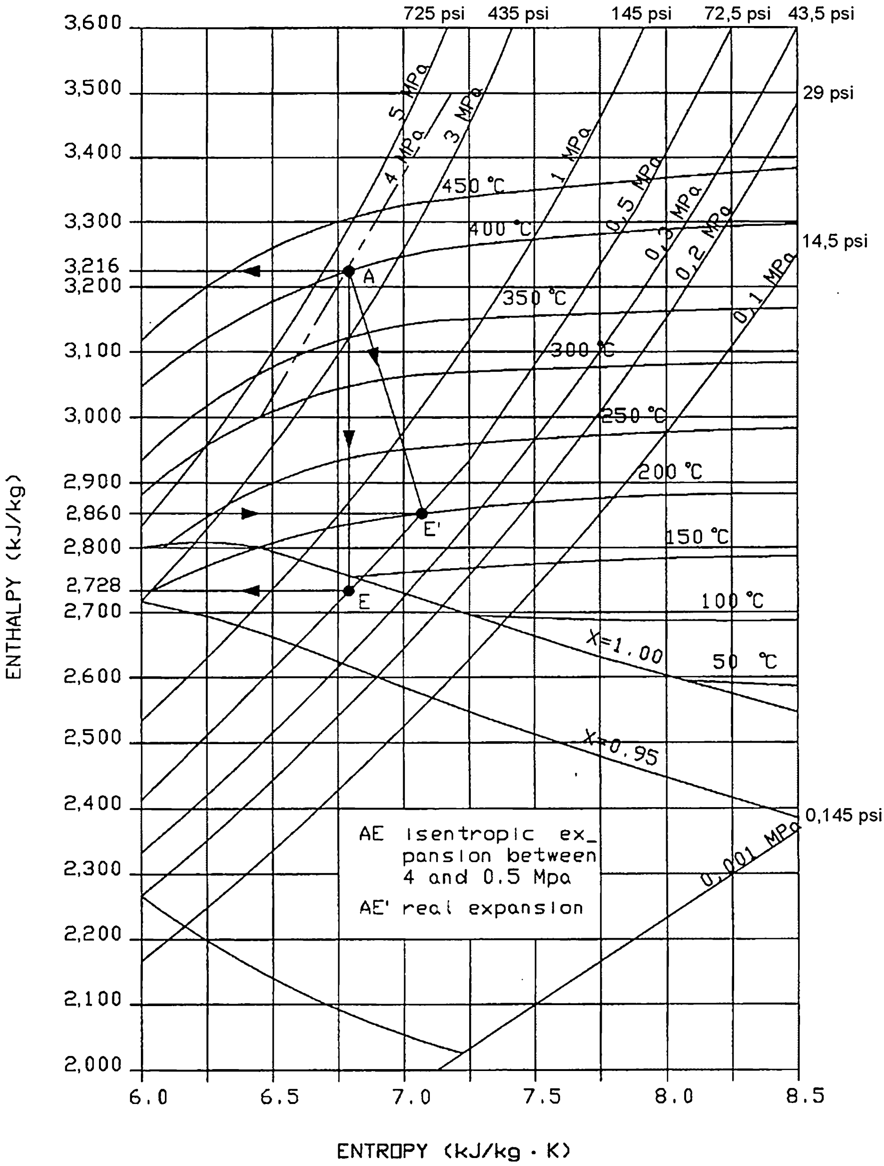


Fig. 9.3 Theoretical steam rate and available energy representation by the Mollier diagram

$$\text{TSR (kg/kWh)} = 3{,}600\ \text{(kJ/kWh)}/\text{AE (kJ/kg)}$$

where, 3,600 kJ/kWh = conversion factor (see Table 2.4) AE = $h_{in} - h_{out}$ along an isentropic expansion, h_{out} = enthalpy of the steam at outlet condition, h_{in} = enthalpy of the steam at inlet condition.

Table 9.2 Theoretical steam rate

	MPa	0.034	0.172	0.345	0.690	1.034	1.379	0.000333
Exhaust steam pressure	psi	5	25	50	100	150	200	0.048
Case 1 (Input 1.034 MPa = 150 psi)								
TSR values	lb/kWh	21.7	31.1	46.0				10.88
	kg/kWh	9.8	14.1	20.9				4.9
AE values	kJ/kg	365.74	255.19	172.53				729.475
Case 2 (Input 1.724 MPa = 250 psi)								
TSR values	lb/kWh	16.6	21.7	28.2	45.2	76.5		9.34
	kg/kWh	7.5	9.8	12.8	20.5	34.7		4.2
AE values	kJ/kg	478.11	365.74	281.44	175.59	103.74		849.752
Case 3 (Input 2.758 MPa = 400 psi)								
TSR values	lb/kWh	13.0	16.0	19.4	26.5	35.4	48.2	8.04
	kg/kWh	5.9	73	8.8	12.0	16.1	21.9	3.6
AE values	kJ/kg	610.51	496.04	409.10	299.49	224.20	164.66	987.150
Case 4 (Input 4.137 MPa = 600 psi)								
TSR values	lb/kWh	11.1	13.2	15.4	19.4	23.8	29	7.25
	kg/kWh	5.0	6.0	7.0	8.8	10.8	13.2	33
AE values	kJ/kg	715.01	601.26	515.36	409.10	333.47	273.67	1,094.715
Case 5 (Input 5.861 MPa = 850 psi)								
TSR values	lb/kWh	9.8	11.5	13.1	15.9	18.6	21.5	6.72
	kg/kWh	4.4	5.2	5.9	7.2	8.4	9.8	3.0
AE values	kJ/kg	809.86	690.14	605.85	499.16	426.70	369.14	1,181.054
Case 6 (Input 8.619 MPa = 1,250 psi)								
TSR values	lb/kWh	8.8	10.1	11.3	13.3	15.1	16.8	6.26
	kg/kWh	4.0	4.6	5.1	6.0	6.8	7.6	2.8
AE values	kJ/kg	901.89	785.81	702.36	596.74	525.60	472.42	1,267.840
Case 7 (Input 9.998 MPa = 1,450 psi)								
TSR values	lb/kWh	8.4	9.5	10.5	12.2	13.8	15.2	6.01
	kg/kWh	3.8	4.3	4.8	5.5	63	6.9	2.7
AE values	kJ/kg	944.84	835.44	755.87	659.54	575.12	522.15	1,320.579
Initial steam condition								
		Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
MPa		1.034	1.724	2.758	4.137	5.861	8.619	9.998
psi		150	250	400	600	850	1,250	1,450
°C		186	260	343	399	441	482	510
°F		366	500	650	750	825	900	950
kJ/kg		2,781	2,935	3,105	3,209	3,281	3,346	3,399
Btu/lb		1,196	1,262	1,335	1,380	1,411	1,438	1,461

Typical values of TSR are 7–8 kg of steam/kWh with a pressure drop from 4 MPa (580 psi) to 0.4 MPa (58 psi).

The previous TSR value can conveniently be converted into an Actual Steam Rate (ASR) by introducing the efficiency of the turbine, which takes into account the shift from the isentropic expansion and the efficiency of the electric generator. Then:

$$\text{ASR (kg/kWh)} = \text{TSR}/(\eta_T \times \eta_G) = (3{,}600/\text{AE}) \times 1/(\eta_T \times \eta_G)$$

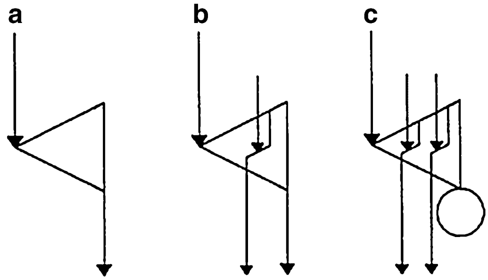


Fig. 9.4 Steam turbine types for cogeneration systems: (**a**) straight non-condensing (**b**) single extraction non-condensing (**c**) double extraction condensing

where, η_T = turbine efficiency, η_G = electric generator efficiency from the shaft to the electric output section.

The electric output is then the ratio between the Actual Steam Flow (ASF) through the turbine as required by the process and the ASR:

$$P_e \text{ (kW)} = \text{ASF (kg/h)/ASR (kg/kWh)}$$

Typical values are 10,000 kg/h of steam to produce 1,000 kW of electric power with a pressure drop of 4 MPa (580 psi).

If ASR is established, depending on the turbine cycle and operating characteristics, the relationship between the inlet and outlet steam enthalpy is as follows:

$$h_{out} = h_{in} - \text{AE} \times \eta_T = h_{in} - 3{,}600/(\eta_G \times \text{ASR}) \text{ (kJ/kg)}$$

Depending on the size of the turbine, on the quantity and quality of process steam demand and other operating factors, several options are available (see Fig. 9.4): straight non-condensing turbine, single or multiple extraction non-condensing turbine.

Figure 9.5 illustrates a typical industrial steam turbine cycle. It consists of a high-pressure boiler, generally 4–10 MPa (580–1,450 psi) generating superheated steam for admission to a backpressure or non-condensing steam turbine. The steam turbine drives either an electric generator or other equipment such as compressors, pumps, etc. The majority of the steam energy content remains in the outlet steam which will be utilized in the process; the energy required for mechanical power and related losses is delivered between the inlet and outlet of the turbine.

The main factors governing the optimal exploitation of a steam turbine cycle based on a fixed exhaust pressure and a constant net heat to process can be summarized as follows:

- The overall efficiency of a steam turbine plant is influenced by the inlet volume flow, inlet–outlet pressure ratio, geometry associated with turbine staging, throttling losses, mechanical coupling, and electric generator losses;

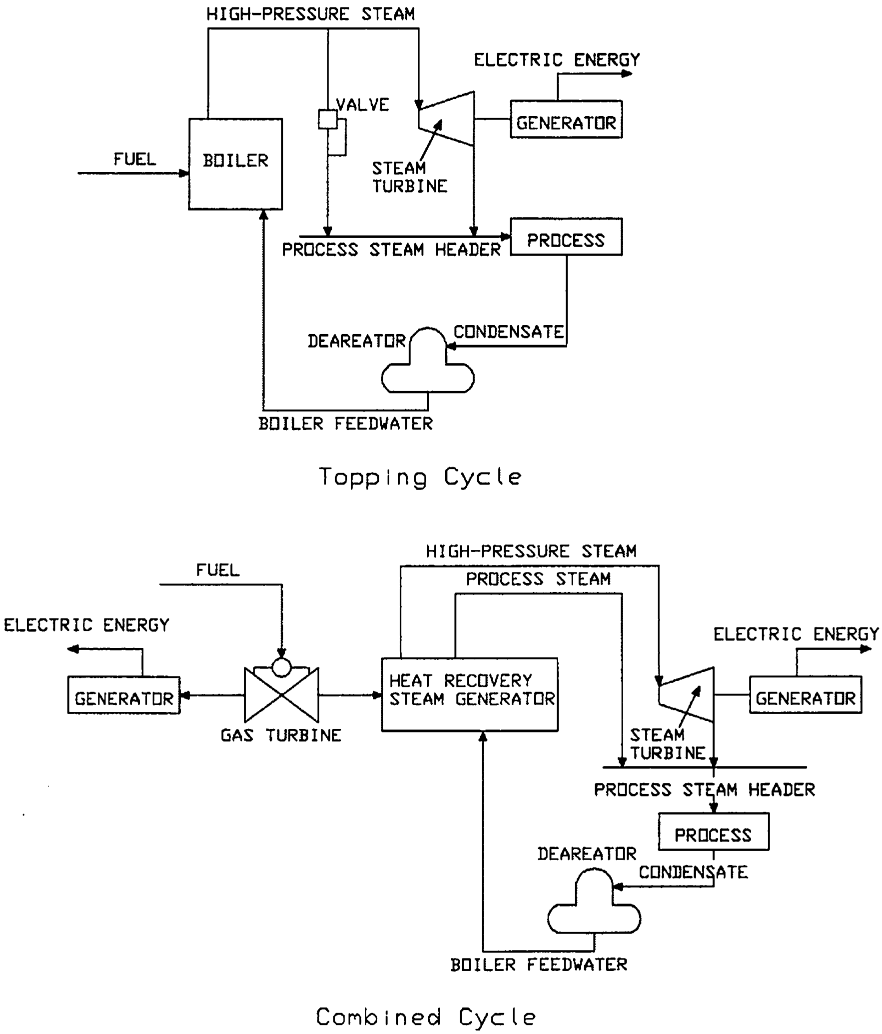


Fig. 9.5 Plant combined cycle cogeneration steam system

- Table 9.3 shows typical values of turbine and overall efficiency and ASR coefficient for a range of unit sizes of backpressure and condensing turbines. Small single-valve, single-stage units have low efficiency, less than 50 %. The multivalve, multistage units may reach an efficiency of up to 80 % in large power plants; in consequence, the greater the efficiency, the greater the electric power generated with the same steam flow. A proper sizing of the prime mover to meet process requirements is the first step in the optimization of a cogeneration plant;

Table 9.3 Overall efficiency and ASR coefficient for backpressure and condensing turbines

Type of unit	Size (MW)	AE (kJ/kg)	Range η_T (%)		Range $\eta_T \times \eta_G$ (%)		Range ASR lb/kWh		Range ASR kg/kWh	
Backpressure	0.1–1	515.36	40	50	38	48	40.5	32.4	18.4	14.7
Single-valve/single-stage	1–5	515.36	65	75	62	71	24.9	21.6	11.3	9.8
Multivalve/multistage	5–25	515.36	75	80	71	76	21.6	20.3	9.8	9.2
Condensing	0.1–1	1,320.579	40	50	38	48	15.8	12.7	7.2	5.7
Single-valve/single-stage	3–20	1,320.579	70	76	67	72	9.0	8.3	4.1	3.8
Multivalve/multistage	2–50	1,320.579	76	80	72	76	8.3	7.9	3.8	3.6

Notes
Typical operating conditions for medium power backpressure turbine
See Case 4 in Table 9.2 with output pressure 0.34 MPa, 50 psi
Input steam 4.137 MPa, 600 psi
Typical operating conditions for condensing turbines
See Case 7 in Table 9.2

- The overall efficiency of a steam turbine diminishes with the output power rate, so there is always a minimum useful value below which operation is not economically worthwhile. Technical characteristics given by manufacturers must be consulted;
- An increase in the initial steam pressure and/or steam temperature will increase the amount of electric energy generated because of the increase in the enthalpy inlet–outlet difference (see Table 9.2). Since an increase of the inlet steam temperatures generally results in an increase of the temperature of the steam supplied to the process, a top limit exists to prevent outlet steam being too highly superheated, as this would necessitate a de-superheating process before delivery to the end users. In such a case, de-superheating water must be added downstream of the turbine outlet; in consequence, the steam flow through the turbine must generally be reduced by the quantity of water added, so that the electric power generated is not increased by using higher inlet steam conditions;
- An increase of the energy available for power generation is also possible if the outlet pressure is reduced with a given set of initial steam conditions. The outlet pressure must be the lowest value compatible with the end user needs;
- An increase in the power generated, with the same fuel consumption in the boiler plant, is possible if feedwater is heated by using steam extracted from turbine stages or exhausted from the process. The level of the power increase depends on the number of heaters and on the temperatures;
- To avoid damage like the erosion of the turbine blades by liquid droplets, inlet steam is always superheated. The outlet steam has generally a quality index of not less than 90 % (see quality index in Sect. 6.4).

Typical values for industrial applications with an electric power ranging between 500 and 5,000 kW are: inlet pressure 4–10 MPa (580–1,450 psi), outlet pressure 0.3–1 MPa (43.5–145 psi), ASR 10–15 kg of steam/kWh, oil consumption in the boiler plant 0.7–1 kg of oil/kWh.

An example of a backpressure steam turbine cogeneration plant is given in Sect. 9.7.

9.4 The Gas Turbine

The gas turbine as prime mover associated with a heat recovery boiler or with direct use of the exhausts in the process is another highly efficient topping cycle. It is available for both mechanical and electric generator drives in a wide range of sizes from a few hundreds to hundreds of thousands of kW. One can distinguish between industrial turbines, in the range from 1 to 400 MW, and aero-derivative turbines in the range from 2 to 40 MW. The revolution speed is generally 3,000–3,600 r/min for power higher than 60 MW and 6,000–12,000 r/min for lower power.

Gas turbine power plants may operate on either an open or a closed cycle, generally the Brayton cycle, as shown in Fig. 9.6. In the open cycle, which is the commoner, atmospheric air is continuously drawn into the compressor; then, air at high pressure enters a combustion chamber where it is mixed with fuel and combustion occurs resulting in combustion exhaust at high temperature. The combustion products expand through the turbine and are discharged into the environment. Part of the mechanical power (typically 60 %) is generally used to drive the compressor and the remainder to generate electricity or to drive other loads.

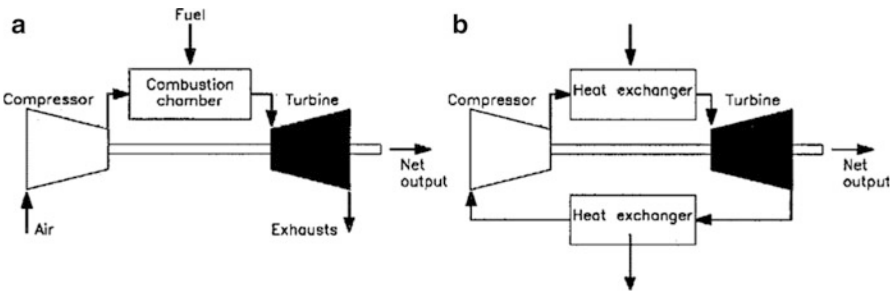


Fig. 9.6 Simple gas turbine: (**a**) open to the atmosphere; (**b**) closed

Figure 9.1 represents a cogeneration plant with a gas turbine if efficiency coefficients from Table 9.1 are used.
The maximum mechanical power available at the shaft does not generally exceed 30–32 % of the turbine input power as fuel in small and medium-sized units (1–5 MW). The amount of recoverable heat depends on the bottom temperature level required by the end user and in consequence on stack-gas temperature.
Typical input power values, in the case of natural gas input, are 350 Sm^3/h (3,330 kW) to produce 1,000 kW of electric power and 1,750 kW of recoverable heat. Specific consumptions, Sm^3 of natural gas per kWh, are lower for bigger gas turbine units.

The overall efficiency of the system, that is, the ratio of total output to input power, is roughly 70–80 %, depending on the bottom level of the thermal energy user which determines the temperature of the stack exhausts. Higher values are not generally possible, because of the great amount of airflow and so of the exhausts. Gasoil instead of natural gas can also be used with appropriate burners.

Combined cycles, where exhausts with or without supplementary burners produce steam and then electric energy in a steam turbine, can also be used for sizes higher than 20 MW of total electric output. These cycles allow for much higher power-producing capability per unit of steam than the backpressure system or the gas turbine by themselves.

Combined cycles with gas turbine and condensing or backpressure turbines are widely used with the ratio of output electric power to input power equal to 46–60 %.

The main data to take into account for selecting gas turbines are as follows:

- Unit fuel consumption/output power. Typical values range between 3 and 3.5 kW input (0.31–0.37 Sm^3/h of natural gas) per 1 kW of output shaft power. Gasoil can be used instead of natural gas;
- Exhaust flow and temperature values, on which the selection of the recovery system can be based;
- The bottom level of temperature required from process end users that limits the temperature at which the exhaust can be cooled;
- Ambient temperature on which depend the density, and so the mass of air flowing through the compressor which is a nearly constant volume flow-rate machine, as well as the bottom temperature of the cycle;
- Atmospheric pressure which varies according to the altitude. In consequence, the density and so the mass of air flowing through the compressor changes;
- Pressure drops upstream the compressor and turbine system and downstream the turbine that may influence the gas turbine useful output;

- The inlet air compression ratio and temperature, to which efficiency is closely related. Air compression ratios of 5–7 are quite common; higher ratios can be used to improve efficiency. Note that the influence of air compressor inlet temperature and of altitude is considerable since they affect the mass of air (and oxygen) available for combustion. Standard conditions for land-based systems are generally 288 K (15.6 °C; 60°F) intake at 0.1 MPa pressure (14.5 psi). A reduction in ambient air density, due to either altitude or high temperature, results in a de-rating of the turbine power output. Typical deratings are as follows: 10 % per 1,000 m of increase of altitude; 2 % per 977 Pa (100 mm H_2O; 3.9 in. H_2O) due to pressure drops downstream and upstream of the turbine; 7–10 % per 10 K or °C (18°F) of increases in ambient temperature;
- The mass of air is considerably higher than the theoretical value needed for combustion (the ratio between input combustion air and natural gas volume is 30–40) since it is imposed by the operating constraints of the air compressor-turbine system;
- If natural gas is used, the pressure of the gas which is roughly 1.5 the air pressure at the inlet of the turbine, must be at least 1–1.2 MPa (145–174 psi). In fact, typical values of air pressure are 0.6–1.5 MPa (87–217.6 psi) for industrial turbines and 1.5–3 MPa (217.6–435 psi) for aero-derivative turbines. A natural gas compressor may be required to ensure these conditions.

With steam production, heat-recovering steam generators are installed downstream of the turbine exhaust outlet. The exhaust temperature is about 723,823 K (450–550 °C; 842–1,022°F) with an excess air of 400 %. The exhaust is 15–18 % oxygen. Turbine exhausts are bypassed if they are in excess.

According to the classification made in Chap. 6, the generators may be unfired and supplementary fired.

Gas turbine exhausts can also be used directly in dryers (tiles, bricks, cereals, other productions).

Unfired generators use the exhaust of the gas turbine unit and they are convective heat exchangers.

Supplementary fired generators use a supplementary burner located downstream of the gas turbine duct to raise the temperature of the exhaust, which can be used as combustion air for it still contains a great quantity of O_2, to a maximum of 1,088–1,143 K (815–870 °C; 1,500–1,600°F). Superheated steam at high pressure is produced, which is suitable for steam turbines in combined cycle plants.

The energy recovery from the exhausts is closely related to the saturation temperature of the steam required by the process, which affects the minimum temperature value at which the exhaust can be cooled through the unfired boiler. Lower values can be obtained if an economizer is installed to pre-heat feedwater (see Sect. 6.9).

A preliminary evaluation of the amount of steam that can be generated is as follows:

$$m_{\mathrm{steam}} = m_{\mathrm{g}} \times c_{\mathrm{p}} \times \left(t_{\mathrm{g}} - t_{\mathrm{s}}\right)/(h_{\mathrm{s}} - h_{\mathrm{i}}),$$

Where, m_{steam} = steam flow-rate produced by the recovery system in kg/s (lb/s), m_g = exhaust gas flow-rate in kg/s (lb/s), c_p = specific heat of the exhaust; typical value is 1 kJ/kg K (0.24 Btu/lb °F), t_g = temperature of the exhaust gas (°C, °F), t_s = saturation temperature of the steam (°C, °F), h_s = enthalpy of the superheated steam or of the saturated steam (kJ/kg, Btu/lb), h_i = enthalpy of the saturated liquid in the steam drum (kJ/kg, Btu/lb).

In the case of supplementary burners, the same relationship can be used by varying the inlet gas temperature until the desired steam flow is reached.

Typical values of operating parameters for several gas turbines are shown in Table 9.4. Notice that the output power can be regulated by varying the input fuel or by throttling the air input in the compressor.

Additional equipment to start the compressor, such as electric motor, reciprocating engine motor, or compressed air tanks, is always required.

An example of a gas turbine cogeneration plant is given in Sect. 9.7.

9.5 The Reciprocating Engine

Reciprocating engine types, principally the spark-ignited gas engine for natural gas or the Diesel engine for liquid fuel, are widely applied in cogeneration systems to drive electric generators and mechanical loads such as compressors and pumps.

Engines are available in a wide range of power, from several to thousands (kW) at different operating speeds ranging from 100 to 1,800 r/min according to the size and technical characteristics of the system.

Figure 9.1 represents a cogeneration plant with a reciprocating engine if efficiency coefficients from Table 9.1 are used. Notice that heat recovery (50 %) is usually shared in high temperature exhaust (20 %) from which steam can be produced and hot water from cooling (30 %).

Exhaust gas, at a temperature in the range of 623–723 K (350–450 °C; 662,842°F), permits steam generation at a saturation pressure of 0.3–1 MPa, suitable for industrial applications.

Jacket and piston water cooling as well as lubricant cooling water can provide hot water at an average temperature of 343–353 K (70–80 °C; 158–176°F), which can be used for space-heating or industrial low-temperature end users. Superheated water can also be produced thanks to specially designed engine and recovery system.

The maximum mechanical power available at the shaft generally ranges from 40 to 45 % of the engine input power as fuel. The higher values refer to turbocharged machines. The amount of recoverable heat, roughly 45–40 %, depends on the bottom temperature level required by the end user in the form of hot water.

Table 9.4 Technical parameters for standard gas turbines

Electrical size unit	Input power	Specific consumption		Input fuel as natural gas		Input air flow			Heat recovery	Efficiency		
										Heat recovery	Electrical	Total
MW	MW	kJ/kWh	Btu/kWh	10^6 Btu/h	Sm^3/h	10^3 Sm^3/h	10^3 kg/h	10^3 lb/h	MW	%	%	%
0.6	2.7	16,364	15,511	9.3	286.0	12.9	16.6	36.6	1.4	50	22	72
1	4.0	14,400	13,649	13.6	419.5	18.9	24.4	53.7	2.0	50	25	75
5	15.6	11,250	10,664	53.3	1,638.7	73.7	95.1	209.7	7.8	50	32	82
10	31.3	11,250	10,664	106.6	3,277.5	147.5	190.3	419.5	15.6	50	32	82
25	65.8	9,474	8,980	224.5	6,900.0	310.5	400.5	883.1	32.9	50	35	88
40	114.3	10,286	9,749	390.0	11,986.3	539.4	695.8	1,534.0	57.1	50	35	85
100	263.2	9,474	8,980	898.0	27,600.0	1,242.0	1,602.2	3,532.2	131.6	50	38	88
200	526.3	9,474	8,980	1,796.0	55,199.9	2,484.0	3,204.4	7,064.4	263.2	50	38	88

Notes

Air density in standard conditions 1.29 kg/Sm^3 (0 °C, 32°F; 0.1 MPa, 145 psi)

Combustion air 45 Sm^3 per unity of Sm^3 of natural gas (standard for natural gas 15.6 °C, 60°F)

Air pressure ratio in the range 5–7

Exhaust temperature 723 K, 450 °C, 842°F; stack exhaust temperature 423 K, 150 °C, 302°F

Temperature drop available for recovery 300 K, 300 °C, 540°F

Cooling equipment must also be provided if users of hot water are not in operation. Exhausts will be discharged into the atmosphere if they are in excess.

Typical input-power values, in the case of natural gas input, are 250 Sm^3/h (2,400 kW) to produce 1,000 kW of electric power and 1,100 kW of recoverable heat. The overall efficiency of the system, that is, the ratio between output and input power, is roughly 85 %. The ratio of output electric power to input power is roughly 40 %.

Typical values of operating parameters for several reciprocating engines fed by natural gas are shown in Table 9.5. As a general rule, this type of prime mover is attractive if the process requires large quantities of low-level heat recovered from the jacket water and lubricant oil cooling systems which may reach half of heat rejection. An example of a reciprocating engine is reported in Sect. 9.7.

9.6 Determining the Feasibility of Cogeneration

Cogeneration feasibility is based on economic and technical factors, which have to be correlated to one another to complete a valid evaluation. Major factors for consideration are as follows:

- The ratio between electricity and fuel site demand, defined as daily, monthly, and yearly ratio. This ratio must be consistent with the ratio between electric output power and heat recovery for the chosen cogeneration system;
- The profile of heat demand, including temperature levels of end user requirements and typical fluctuations of the demand (hourly, daily, monthly, yearly). Temperature levels must be consistent with the level of heat rejected from the cogeneration system;
- The profile of electric demand and typical fluctuations as for the thermal profile. Thermal and electric profiles must be correlated with each other;
- Purchased fuel and electricity costs, present and projected future costs;
- Working hours per year and per total life of the plant;
- Plant system sized for present site needs and for the future;
- Capital cost of the cogeneration plant and operating cost during the life of the plant;
- Environmental issues.

Many cogeneration approaches can be followed in order to make a choice among system types and sizes. However, in order to ensure the highest efficiency of the system, the recovery of the rejected heat must be effective in any operating condition of the cogeneration plant. Additional boiler plants will satisfy the end user requirements, if these are higher than the recovery heat. Depending on the industrial processes, this constraint can be more or less important in determining the size of the plant.

Table 9.5 Technical parameters for reciprocating engines

Electrical size unit	Input power	Specific consumption		Input fuel as natural gas		Input air flow			Heat recovery	Efficiency		
										Heat recovery	Electrical	Total
MW	MW	kJ/kWh	Btu/kWh	10^6 Btu/h	Sm^3/h	10^3 Sm^3/h	10^3 kg/h	10^3 lb/h	MW	%	%	%
0.25	0.7	9,474	8,980	2.2	69.0	1.0	1.3	2.9	0.3	45	38	83
1	2.4	8,571	8,125	8.1	249.7	3.7	4.8	10.7	1.1	45	42	87
5	11.9	8,571	8,125	40.6	1,248.6	18.7	24.2	53.3	5.4	45	42	87
10	23.8	8,571	8,125	81.2	2,497.1	37.5	48.3	106.5	10.7	45	42	87

Notes
Air density in standard condition 1.29 kg/Sm^3 (0 °C, 32°F; 0.1 MPa, 145 psi)
Combustion air 15 Sm^3 per unity of Sm^3 of natural gas (standard for natural gas 15.6 °C, 60°F)

A first approach is designing a system which is capable of meeting thermal load requirements, regardless of electricity demand. It is connected to the utility grid and sells excess or buys additional electricity depending on the site's thermal and electric profile and on the operating conditions.

A second approach is designing a system capable of meeting either peak or base electric load requirements, regardless of the thermal demand, which nevertheless must be greater than the heat rejected. It is connected to the utility grid and sells excess or buys additional electricity depending on the sizing and on the operating conditions.

A third approach is designing a system independent of the utility grid. It requires overcapacity or redundant equipment to ensure reliability, which is guaranteed by the utility in the first two approaches. These systems have traditionally been oversized to meet peak electric demand, with supplementary equipment to satisfy the thermal demand if necessary.

9.7 Practical Examples

Three examples of cogeneration and one example of trigeneration plants are given below. A comparison of savings in primary energy is also shown.

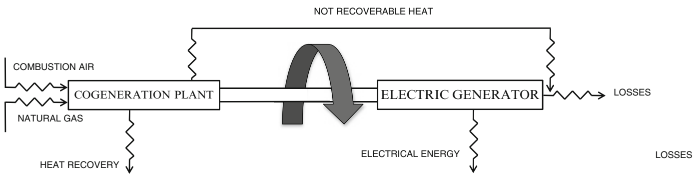
Notice that cogeneration and trigeneration plants allow for a saving in primary energy, not in the energy entering the site. They can achieve an energy-cost saving only if the balance between electric energy and fuel energy costs is favorable for the site.

Example 1 Cogeneration plant with steam turbine

Figure 9.7 shows the energy balance and the economical evaluation of a steam turbine cogeneration plant with 1,000 kW electric power in typical working conditions (input–output pressure drop equal to 3.793 MPa, 550 psi). The heat recovery and electrical efficiency are those shown in Table 9.1. Specific consumptions due to the production of electric energy are calculated as Sm^3/kWh or kg of oil/kWh entering the steam boiler plant.

The figure shows also primary energy saving (TOE), in comparison with standard utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities).

The reference costs of electric energy and fuel are used for the economic evaluations. A preliminary evaluation can be made on the basis of the average cost of electric energy purchased from utilities (see Table 19.4); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy.



100% F — **74%** H — **13%** E — **13%** L

85% Net heat recovery index (1)
12.8% Primary Energy Saving index (2)

7,692 kW FF — **5,692** kW — **1,000** kW Pe — **1,000** kW
597 Sm^3/h (3)
806.8 Sm^3/h — **663.3** Sm^3/h (4) = **143.5** Sm^3/h
0.13 U/Sm^3 — **0.143** Sm^3/kWh

1,613 CO_2 kg/h — **1,327** CO_2 kg/h — **286** CO_2 kg/h
0.286 CO_2 kg/kWh
104.9 U/h - **86.2** U/h = **18.7** U/h

cost for fuel	**1.87**	cU/kWh	a
O&M cost	**1.0**	cU/kWh	b
overall production cost	**2.87**	cU/kWh	c=a+b
reference cost from utility	**10**	cU/kWh	d
cost saving	**7.13**	cU/kWh	e=d-c
operating hours	**6,000**	h/year	f
electric energy production	**6,000,000**	kWh/year	g=f*Pe
yearly cost saving	**428,128**	U/year	NACF=e*f (see Chapter 19)
capital cost	**1,200**	U/kW	i
	1,200,000	U	I=i*Pe
payback	**2.8**	year	I/NACF (see Chapter 19)
IRR	**34**	%	(see Chapter 19)
primaryenergy saving	**584**	TOE/year	(5)

IRR based on 10 year of life
input data
(1) H/(H+E) (2) $(1-F/(E/\eta_p+H/\eta_b))*100$ $(5)=(2)*FF*(E/\eta_p+H/\eta_b)*3{,}600*f/41{,}860/1{,}000$
(4)= $(3)/\eta_b$ *where η_p =electrical production and transmission efficiency=40% and η_b =efficiency of the boiler to replace=90%*
natural gas LHV=34,325 kJ/Sm^3 equivalent oil NHV= 41,860 kJ/kg 2 $kgCO_2/Sm^3$ of natural gas

Fig. 9.7 Cogeneration plant with backpressure steam turbine

Local regulations concerning the selling of energy to the utilities and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account, typically a fixed cost per unit of kWh produced.

Example 2 Cogeneration plant with gas turbine

Figure 9.8 shows the energy balance and the economical evaluation of a gas turbine cogeneration plant with 1,000 kW electric power in typical working conditions. The heat recovery and electrical efficiency are those shown in Table 9.1. Specific consumptions due to the production of electric energy are calculated as Sm^3/kWh entering the plant.

The figure shows also primary energy saving (TOE) in comparison with standard utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities).

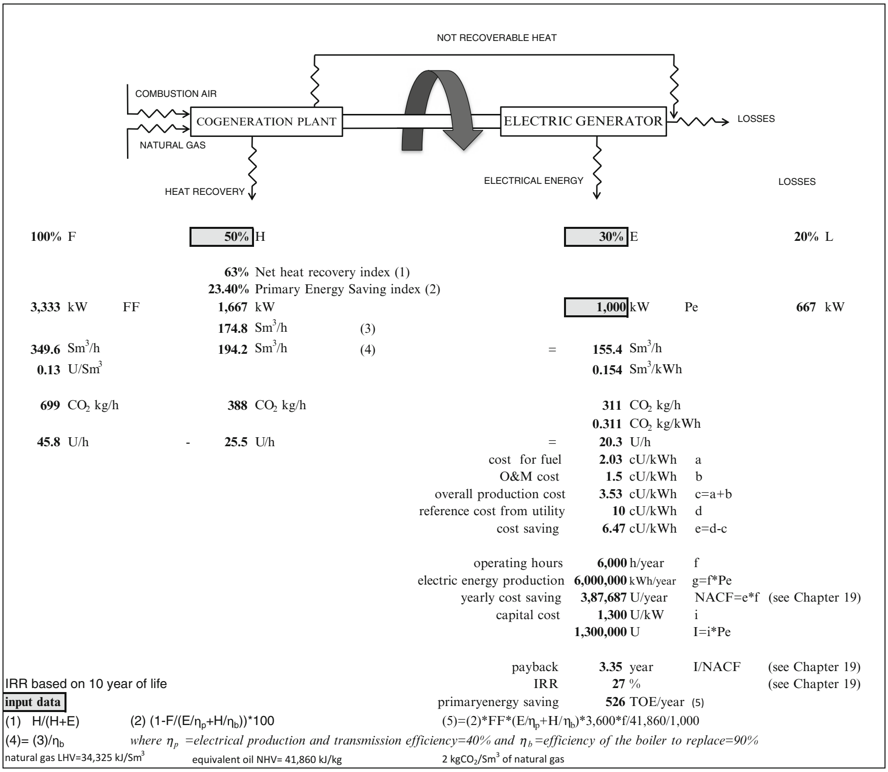


Fig. 9.8 Cogeneration plant with gas turbine

The recoverable heat form flue-gases varies according to end user requirements: hot air for drying, steam and hot water, with or without additional burners.

The reference costs of electric energy and fuel are used for the economic evaluations. A preliminary evaluation can be made on the basis of the average cost of electric energy purchased from utilities (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. Local regulations concerning the selling of energy to the utilities and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account, typically a fixed cost per unit of kWh produced.

Example 3 Cogeneration plant with reciprocating engine

Figure 9.9 shows the energy balance and the economical evaluation of a reciprocating engine cogeneration plant with 1,000 kW electric power in typical working conditions. The heat recovery and electrical efficiency are those shown in

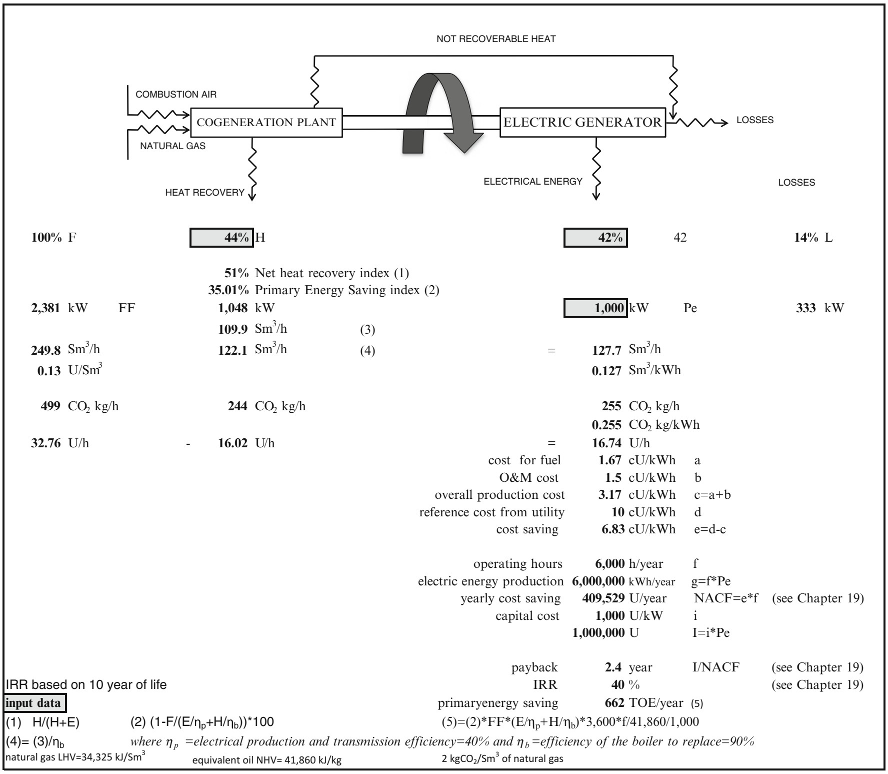


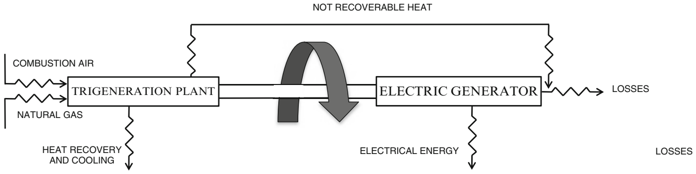
Fig. 9.9 Cogeneration plant with reciprocating engine

Table 9.1. Specific consumptions due to the production of electric energy are calculated as Sm^3/kWh entering the plant.

The figure shows also primary energy saving (TOE) in comparison with standard thermal utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities).

The recoverable heat from flue-gases and engine cooling media varies according to end user requirements: hot air for drying, steam and hot water at different temperatures, to which the possibility of a complete exploitation of the rejected heat is correlated.

The reference costs of electric energy and fuel are used for the economic evaluations. A preliminary evaluation can be made on the basis of the average cost of electric energy purchased from utilities (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. Local regulations concerning the selling of energy to the utility and its purchase from them in emergency must also be considered.



100% F	**44%** H	**42%** E	**14%** L
	51% Net heat recovery index (1)		
	35.01% Primary Energy Saving index (2)		
2,381 kW FF	**1,048** kW (3)	**1,000** kW Pe	**333** kW
248.4 Sm^3/h	**1,048** kW (4)=(3)		
	0.75 COP absorption		
	876 kWcoling		
	4 COP cooling compressor to replace		
	196 kWelectrical compressor to replace		
0.13 U/Sm^3	**0.1** U/kWh	**0.127** Sm^3/kWh	
499 CO_2 kg/h	**0** CO_2 kg/h	**499** CO_2 kg/h	
		0.499 CO_2 kg/kWh	
32.76 U/h	- **19.64** U/h	= **13.12** U/h	

cost for fuel	**1.31**	cU/kWh	a	
O&M cost	**1.5**	cU/kWh	b	
overall production cost	**2.81**	cU/kWh	c=a+b	
reference cost from utility	**10**	cU/kWh	d	
cost saving	**7.2**	cU/kWh	e=d-c	
operating hours	**6,000**	h/year	f	
electric energy production	**6,000,000**	kWh/year	g=f*Pe	
yearly cost saving	**431,283**	U/year	NACF=e*f	(see Chapter 19)
capital cost	**1,200**	U/kW	i	
	1,200,000	U	I=i*Pe	
payback	**2.8**	year	I/NACF	(see Chapter 19)
IRR	**34**	%		(see Chapter 19)
primaryenergy saving	**662**	TOE/year	(5)	

IRR based on 10 year of life

input data

(1) H/(H+E) (2) $(1-F/(E/\eta_p+H/\eta_b))*100$ (5)=(2)*FF*$(E/\eta_p+H/\eta_b)$*3,600*f/41,860/1,000

(4)= (3) *where* η_p *=electrical production and transmission efficiency=40% and* η_b *=efficiency of the boiler to replace=90%*

natural gas LHV=34,325 kJ/Sm^3 equivalent oil NHV= 41,860 kJ/kg 2 $kgCO_2/Sm^3$ of natural gas

Fig. 9.10 Trigeneration plant with reciprocating engine

Maintenance costs, too, must be taken into account, typically a fixed cost per unit of kWh produced.

Example 4 Trigeneration plant with reciprocating engine

Figure 9.10 shows the energy balance and the economical evaluation of a reciprocating engine trigeneration plant with 1,000 kW electric power in a typical working condition. The heat recovery and electrical efficiency are those shown in Table 9.1. Specific consumptions attributed to the production of electric energy are calculated as Sm^3/kWh entering the plant.

The figure shows also primary energy saving (TOE) in comparison with standard thermal utility plants and variation of the site energy consumption (additional fuel consumption and reduction of kWh from utilities). Only recovered heat is taken into account for the primary energy saving (TOE) as for Example 3, regardless the subsequent conversion into cooling.

The recoverable heat from flue-gases and engine cooling media varies according to end user requirements: hot air for drying, steam and hot water at different temperatures, to which the possibility of a complete exploitation of the rejected heat is correlated.

In trigeneration plant heat is converted into cold water by using absorption systems.

The reference costs of electric energy and fuel are used for the economic evaluations. A preliminary evaluation can be made on the basis of the average cost of electric energy purchased from utilities (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. Local regulations concerning the selling of energy to the utility and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account, typically a fixed cost per unit of kWh produced.